TIMKEN

Technical Article

A Tale of Two Greases:

Comparison of Wind Energy Grease Performance Using Bench and Component Testing

TIMKEN

Authors

Douglas R. Lucas
Kuldeep K. Mistry

Table of Contents

A Tale of Two Greases:
Comparison of Wind Energy Grease Performance Using Bench and Component Testing

Abstract

Testing was performed on two typical wind turbine mainshaft bearing greases. The tests were used to compare the performance of these greases in conditions that simulate operation in the field. Previously, Pierman [1] and Mistry [2] compared nearly a dozen wind energy greases by examining them with standard lab tests to identify which greases would likely perform best in various field conditions. Two of these greases were then selected for a more detailed investigation with three tests performed: film thickness, bearing life test, and a torque and temperature rotational test. For the rotational test, the greases were added to a 2MW direct drive wind mainshaft bearing which was operated under preload only. The results of these tests are summarized in this report.

Wind Turbine Mainshaft Bearings

Wind turbine mainshaft bearings are designed to carry the variable loading associated with constantly changing wind speeds and directions in a wide variety of environmental conditions. Lubrication is extremely important to achieve the 20-year-plus design life requirement, even with a properly designed bearing. The relative motion between the rolling element and the race during operation entrains the lubricant to generate a film thickness that separates the steel surfaces. The film thickness must be large enough to prevent interaction between the asperity tips from the surface roughness of the two interacting surfaces.

The lambda ratio is the ratio of the lubricant film thickness divided by the composite surface roughness. Because of the low and variable speeds, it is challenging to achieve a lambda ratio greater than the target of 1.0. Micropitting is the predominant damage mode observed on wind turbine spherical roller mainshaft bearings. Micropitting damage risk increases as the lambda ratio decreases.

Proper lubrication and regular maintenance of mainshaft bearings is very important to achieving a trouble-free bearing. Oil is a popular method to lubricate bearings; however, most wind turbine mainshaft bearings are grease-lubricated. Grease is composed of a thickener, typically a soap, emulsified with an oil. Additives are incorporated into the greases to improve performance in demanding applications. Unlike oil, grease is more viscous and will not readily migrate back into the roller path once it is pushed away. This can lead to less entrained grease in the roller contact and a lower film thickness.

The NLGI grade is a measure of the relative hardness or consistency of a grease and affects grease migration in the bearing. For example, a NLGI Grade 2 grease may have the consistency of peanut butter, whereas a Grade 0 may be like mustard. Under operation, oil bleeds from the thickener to lubricate the bearing, which will eventually lead to thickening of the grease and a degradation of the grease's lubrication properties. Therefore, it is important to regularly relubricate the bearings to replenish the oil and maintain the proper consistency. In wind turbines, relubrication is typically performed manually every six months during regular maintenance or more regularly with an automatic lubrication system.

In contrast to wind turbine gearboxes, which use primarily ISOVG 320 synthetic PAO (polyalphaolefin) oils, the lubrication used on wind turbine mainshaft bearings is not as consistent. Although many of the greases are synthetic lithium or lithium complex greases with an NLGI grade of 1.5 or 2, the base oils may be mineral, a synthetic-mineral blend, or synthetic ester oils. The base oil viscosity varies greatly, ranging from 130 to 680 cSt at 40°C. The oil separation also varies greatly, from 1% to 5.5%. Further, there is a wide array of additive packages used in the greases, which makes it even more challenging to evaluate which grease is best for wind turbine mainshaft bearings.

Previous Studies

In 2013, Dave Pierman published a paper in which he evaluated more than 10 typical wind turbine mainshaft greases [1]. He summarized the importance of specific tests and concluded that lubricant film thickness, oil separation, and bearing wear behavior are three of the most critical grease characteristics, regardless of the wind turbine environment. Obviously, low-temperature torque and grease pumpability are important tests to consider if turbines operate in cold weather conditions (-40°C). Bearing operating temperature is an important measurement that can be used to evaluate the oxidation rate and expected life of the grease. Corrosion protection is required in offshore applications and fretting protection is required to prevent fretting wear and false brinnelling on the raceways during oscillation or stationary events.

Kuldeep Mistry's 2019 paper [2] further evaluated the film thickness, traction coefficients, and bearing temperature, torque and grease migration for several of the greases from the Pierman work. His work confirmed that the greases with higher base oil viscosities resulted in larger film thicknesses. Film thickness refers to the thickness of the oil separating the roller and raceway (see **Figure 1**). An example 3D measurement of this film thickness is shown in **Figure 2**. However, these greases also have higher traction coefficients and operating temperatures.

Testing also showed that the greases with lower base oil viscosity performed much better on the grease migration tests, allowing the grease to flow back into the bearing contact surfaces. This is expected to provide more effective lubrication at the contact surfaces. Additionally, although the lower-viscosity grease had a slightly higher traction coefficient, the bearing torque and temperature measured were significantly lower (35% less torque and 15% lower temperature (-10°C)). This confirms that a lower base oil-viscosity grease with the right chemistry can properly lubricate a bearing, regardless of the potentially lower calculated lambda ratio associated with the lower base oil viscosity.

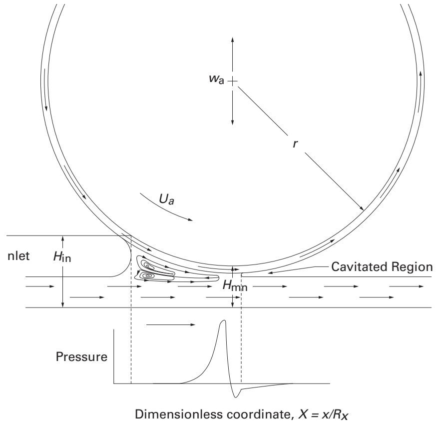

Figure 1: Theoretical 2D representation of ball-disc contact

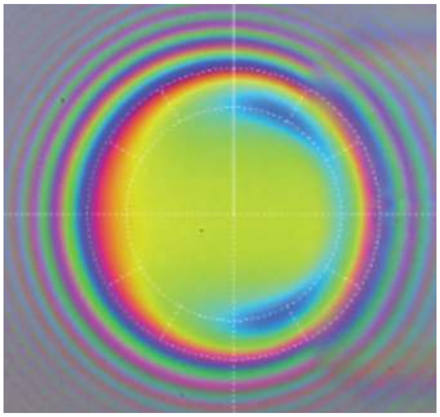

Figure 2: Typical 3D film thickness measurement of ball-disc contact

All the properties discussed here are important, but they cannot be evaluated during wind turbine operation to judge the lubrication effectiveness. Grease samples should be taken at regular intervals to evaluate the change in these properties over time. Bearing temperature is one of the simplest measurements on a wind turbine that can be related to bearing performance. Bearing torque and temperature increase with friction on a damaged or poorly lubricated bearing. It is a generally accepted guideline that every 10°C temperature increase will double the oxidation rate of a lubricant, resulting in shorter (half) grease life.

Test Grease Properties

As a further continuation of the testing from Pierman and Mistry, two typical wind energy mainshaft bearing greases were selected to evaluate the performance in a mainshaft bearing, specifically looking at operating torque and temperature. One grease has a high base oil viscosity and one a low base oil viscosity, which are near the extremes of the ranges used currently. The grease properties are highlighted in **Table 1**.

	Grease A	Grease B
NLGI Grade, DIN 51818	1	1.5
Penetration, Worked, 25°C, ASTM D 217	310 – 340	305
Thickener Type	Lithium complex	Lithium complex
Oil Separation, DIN 51817 7 days @ 40°C, Static, %	4.0	1.5
Base Oil	Semi-synthetic hydrocarbon oil	Synthetic oil
Base Viscosity (cSt @ 40°C) ASTM D 445	130	460
Base Viscosity (cSt @ 100°C) ASTM D 445	14	46.5
Dropping Point, °C, ASTM D 2265	> 250	255
Service Temperature, °C	-40 to 150	-30 to 150

Table 1: Test grease properties

Film Thickness Studies

Film thickness tests demonstrated that operating speed and temperature significantly affect film thickness under constant loading conditions. Testing showed that film thickness was reduced at heightened temperatures but increased at higher operating speeds, which are generally accepted relationships. Throughout the various temperature and speed conditions in this study, it generally held true that higher base oil viscosity resulted in higher film thickness during operation.

The PCS-EHD2 testing rig, which uses a steel ball on a glass disk, was used to compare the film thickness for unworked and worked grease samples to simulate new and used greases, respectively. Typically, performance testing is performed only on fresh, unworked grease, but this study was completed to further understand performance changes for greases that have been used in an application.

To simulate the grease shearing that occurs in applications over time, the study greases were worked per ASTM D1831 for 500,000 revolutions or strokes. Testing revealed that the grease film thickness was similar for unworked (new) and worked (used) greases. Moreover, as expected, higher temperatures resulted in lower film thickness for all samples tested. As shown in **Figure 3**, Grease A resulted in lower film thickness than Grease B both at lower speed (entrainment velocity) (0.1 m/s) and higher speed (1 m/s). This was expected because Grease B has a higher viscosity, which resulted in a greater film thickness formation compared to Grease A.

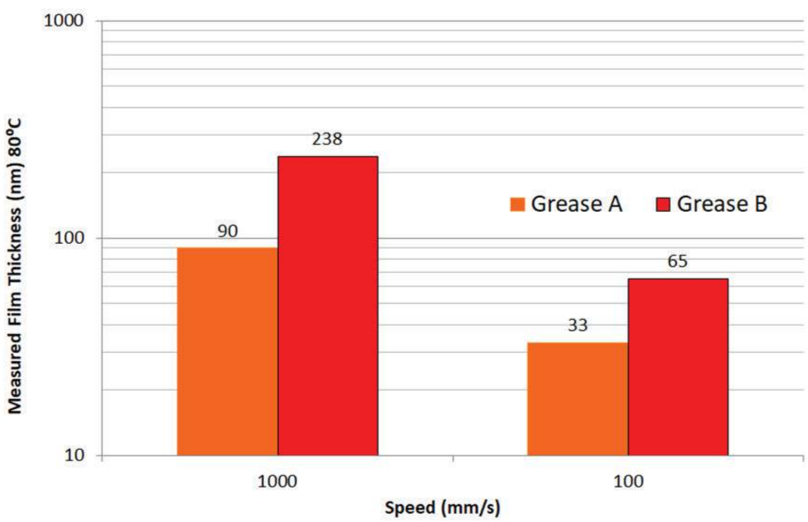

Figure 3: Film thickness of fresh greases tested on the PCS-EHD2 test rig

TIMKEN

Grease-Lubricated Bearing Life Test:

To further evaluate and compare the greases, life testing (first in four) was conducted in a 5" (127 mm) Timken bearing life test rig using 24 M88000 series bearings with special tooling designed to accommodate grease lubrication. The coolant system of the test rig was also modified to use an ISO VG 10 oil instead of water to better maintain and control the temperature of the test bearings to a target cup OD temperature of 160°F. The actual operating temperature for Grease A was approximately 150°F and for Grease B was approximately 170°F.

Following a typical break-in cycle, the life testing was conducted using a constant radial load of 23.3 kN (5240 lbf) at a test speed of 1200 RPM. This loading is the standard life test loading (150% of C90 rating) for this bearing series. The control system was set to shut down if the cup OD temperatures exceeded 210°F. If a high-temperature shutdown occurred, the bearing raceways were inspected for damage. If none was found, the test was still terminated and categorized as a "grease failure," and the data was not included in the final test results. High-temperature shutdown did occur several times with Grease B.

As observed in **Figure 4**, although Grease A is a lower-viscosity base oil grease, it lasted for an average of 97 million revolutions while the higher-viscosity base oil grease, Grease B, had an average life of 36 million revolutions. On average, the bearing life with Grease A was 2.7 times higher than with Grease B. By comparing the 65% confidence bands between the tests, we can state with 90% confidence that Grease A resulted in 1.65 times higher life compared to Grease B when tested under the same conditions. Moreover, both greases survived longer than their predicted lives—on average, Grease A survived more than 5 times longer, and Grease B survived more than 1.25 times longer. The longer grease lives may be linked not only to the actual operating temperatures, but also to the grease additives and thickeners.

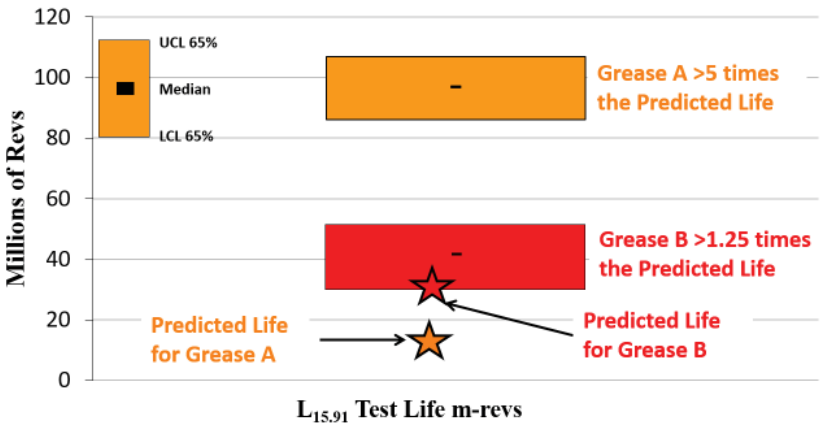

Figure 4: Bearing life of fresh greases tested on the Timken grease-lubricated life test rig

Full-Scale Bearing Testing

Sub-scale tribology test rigs are commonly used to compare lubricant properties. One example used by The Timken Company is the LEM (lubricant evaluation machine), which applies thrust to a tapered roller bearing. The LEM is a great cost-effective option to compare lubricants quickly on smaller bearings. However, for the testing summarized in this paper, a custom test rig was utilized to rotate a wind turbine mainshaft bearing under a preloaded condition. The test bearing NP816906-90WA1 is a double-row tapered roller bearing in a direct mount configuration (see **Figure 5**) in which the bearing effective centers are directed toward each other and overlap in an "X" configuration. This bearing is commonly referred to as a tapered double inner (TDI) bearing and has a one-piece inner ring and two outer rings.

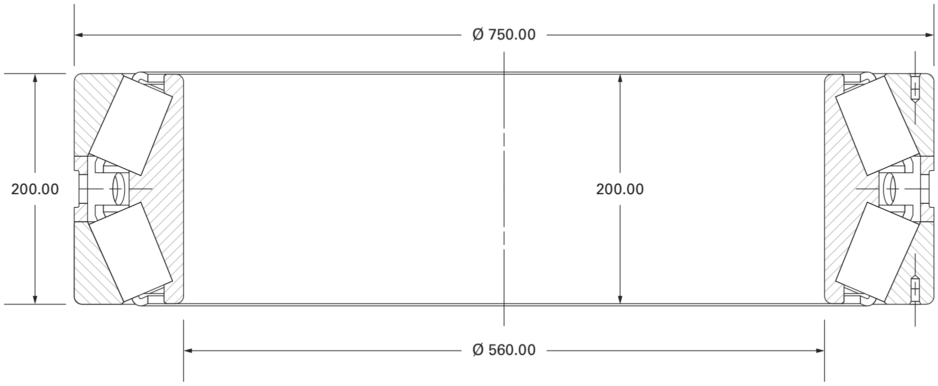

Figure 5: NP816906-90WA1 bearing cross-section and basic dimensions

For this test, the bearing was run with a horizontal centerline, similar to a mainshaft application (see **Figure 6**). The TDI, without a cup spacer, was clamped between two non-rotating plates (figure reference numbers 6, 12). The blind-end plate (12) was floating and the drive-end plate (6) was fixed. Bolts (9) were used to mount and center the bearing outer rings (cups), which were fixed. The inner ring rotated and was piloted and clamped in a fixture connected to the coupling and motor. Four (4) 100-ton hydraulic cylinders (10), as shown in yellow in **Figure 7**, were mounted between the pivoting fixture (1) and the floating cup clamp plate (12). These cylinders were used to apply preload to the bearing assembly. The blue arrows in **Figure 7** show the load path from the cylinders through the bearing assembly. A photo of the fully assembly test rig and assembly is shown in **Figure 8**.

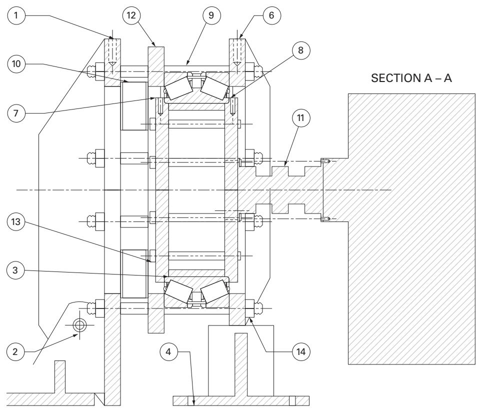

Figure 6: Cross-section of test setup

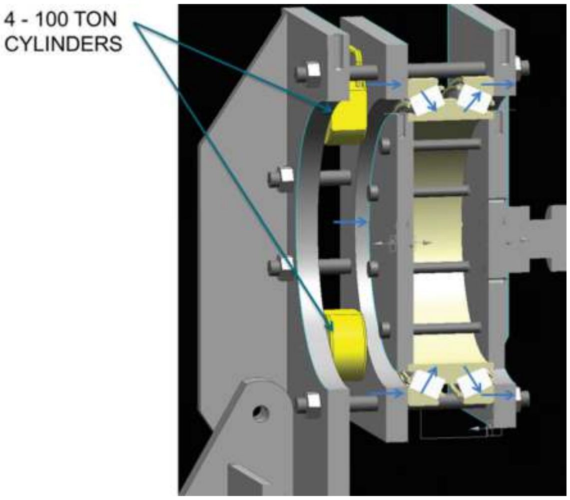

Figure 7: 3D cross-section of test setup showing load path (blue arrows)

Figure 8: Actual test rig

Because the bearing was not sealed, a minimal quantity of grease was used for each test. Three quantities of grease were used: 0.45 kg (1 lb.), 0.90 kg (2 lb.), and 1.35 kg (3 lb.) of grease per bearing row. The bearing was then operated at 15 RPM under an axial thrust load of 1100 kN (247,000 lbf). This load was 135% of the bearing thrust capacity rating at 90M revolutions (Ca90) (35% of Ca1) and resulted in a bearing contact stress of 1500 MPa (218 ksi). The same two greases previously discussed were selected for this test. The grease properties are shown in **Table 1**. These greases were selected because they are commonly used greases in the wind industry and because the grease properties vary widely from each other.

The test was composed of a 30-minute break-in cycle at 15 RPM and a 105kN (23,600 lb.) load. After the break-in cycle, the axial clamping load was applied, and the test was started and run for 90 minutes. A summary of the test duty cycle is shown in **Table 2**.

Step	Load (kN)	Speed (RPM)	Time (minutes)	Grease Quantity (kg)
Break-in	105	15	30	
Test 1	1100	15	90	0.45
Test 2	1100	15	90	0.90
Test 3	1100	15	90	1.35

Table 2: Test duty cycle

Motor torque and bearing temperature were measured and recorded during this testing. Although motor torque is not exactly representative of the actual bearing torque, the torque difference correlates to the lubrication differences in these tests.

The bearing temperature was measured at the center of each outer ring outside diameter (OD) (see **Figure 9**). Ambient air temperature was also measured and was used to normalize the charts to eliminate the effect of ambient temperature on the results.

Figure 9: Location of thermocouples on cup outside diameter

Torque and Temperature Results

Test 1

Test 1 was run using 0.45 kg (1 lb.) of grease per bearing row. At break-in, there was a noticeable temperature difference between the two greases. The temperature of Grease A only increased about 2°C, whereas Grease B increased 5°C. After the break-in, when the axial load was applied, the bearing torques and temperatures were significantly different. Grease A temperature increased to only 10°C above ambient air temperature, whereas Grease B increased to 30°C above ambient temperature.

There was a similar increase in the torque as well. Grease B torque was 5 times that of Grease A (average of 294 N-m (2600 in-lbf) versus 60 N-m (530 in-lbf)). When grease was added at 55 minutes, the torque dropped. There was a slight temperature increase and large torque increase at 80 minutes for Grease A, which was later attributed to grease starvation at the blind end (BE) bearing. When grease was re-added to the bearing, the torque reduced to its original levels.

Test 2

Test 2 used 0.90 kg (2 lb.) of grease per bearing row. Similar to Test 1, the initial bearing temperatures for Grease B during the break-in period were higher. This was expected since Grease B has a higher viscosity and higher NLGI grade than Grease A, both characteristics helping to resist the rolling element's rotation in the bearing. This generated higher friction and temperature.

It was also observed that the bearing temperatures for Grease B reduced with additional grease, whereas the bearing temperatures for Grease A remained the same. This supports the observation that there was inadequate grease used in Test 1 for Grease B, which led to grease starvation, higher friction, and higher temperatures. Although the Grease B bearing temperature was lower for Test 2 by nearly 10°C, it was still higher than Grease A for both tests by 15°C. This temperature differential between the greases is in line with the observations from Mistry [2].

The torque using Grease B dropped to an average of 169 N-m (1500 in-lbf), whereas the Grease A torque stayed low at about 68 N-m (600 in-lbf), which is still only 40% of Grease B. When the test reached 66 minutes, there was an increase in the torque of nearly 79 N-m (700 in-lbf), which is related again to grease purge and the start of grease starvation in the bearing.

Test 3

In the final test, Test 3, the bearings were lubricated with 1.35 kg (3 lb.) of grease per bearing row. The torque and bearing temperatures were similar in Tests 2 and 3. This confirmed the theory that Test 1 was operated with insufficient lubrication. The consistent results from Tests 2 and 3 reveal that the operating temperature of Grease A is 15°C less than that of Grease B.

Also in Test 3, there was an increase in torque between 65 and 75 minutes for Grease B. Adding grease to the bearing at 87 minutes resulted in an increase in torque. This confirmed that the increase in torque observed between 65 and 75 minutes was a result of grease migration within the bearing interfering with the rolling element rotation. The torque for Grease A did increase at 82 minutes but reduced when grease was reintroduced at 88 minutes. In this case, there was inadequate lubrication because of grease purge, which resulted in the torque increase.

A comparison of the grease film thickness of the two greases was made using an average ambient temperature of 25°C. At a 35°C operating temperature, Grease A had a calculated central film thickness of 0.2 um (8 uin). Grease B had a central film thickness of 0.43 um (17 uin) when operating at 50°C. Due to the base oil viscosities being very different (460 versus 130 at 40°C), the resultant calculated film thickness of Grease A was only 45% of Grease B's at the actual operating bearing temperatures.

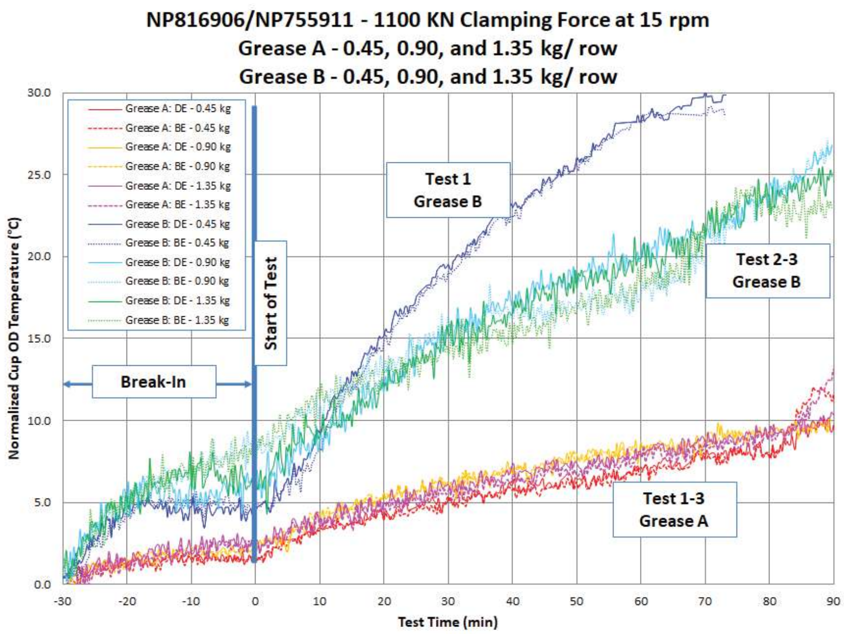

Figure 10: Normalized cup OD temperatures for all tests and both greases

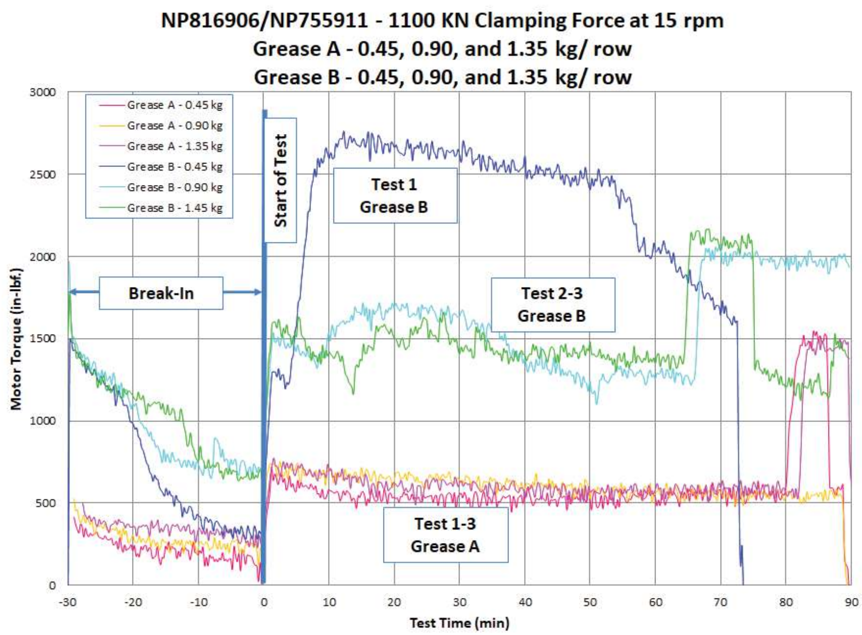

Figure 11: Torque for all tests and both greases

Conclusions

Based on this testing, one can observe that the quantity, type, and consistency of the grease is important to the operating torque and temperature of wind energy mainshaft bearings. Additional conclusions are:

- Although the predicted and measured film thickness for Grease A was lower than Grease B, the life test and temperature and torque testing both showed that Grease A outperformed Grease B.
- Grease A performed much better than expected based on lambda ratios predicted by only the base oil viscosity and lower temperatures. This may be attributed to additives or grease thickener effects, which result in larger lambda ratios and higher actual bearing lives.
- Insufficient grease results in high torque and temperatures, which are many times higher than a bearing that is properly lubricated. Adding grease to the bearing reduces the temperatures.
- When the proper quantity of grease is used, the base oil viscosity and NLGI grade play a critical role in the operating torque and temperature. A viscous grease with a high base oil viscosity operates at a higher temperature—up to 10°–15°C hotter for a slow rotating wind mainshaft bearing—than a less viscous grease with a lower base oil viscosity.
- The torque and temperature results from a full-size bearing are directionally the same as the results from sub-scale bearing testing.
- In this test example, when properly lubricated, Grease A operated with 40% of the torque and 15°C lower temperature compared to Grease B, which is more viscous. The lower temperatures result in a slow oxidation rate, which should lead to a longer grease life.
- Although the lower-viscosity grease (Grease A) calculated a lower lambda ratio and measured lower film thickness, it can still result in improved performance in application.

References

[1] D.A. Pierman. "Main Bearing Lubrication for Wind Turbines – A Systematic Approach for Grease Selection." 25th ELGI Annual General Meeting 2013.

[2] K. Mistry, D. Lucas, and P. Shiller. "Grease Selection for Main-shaft Bearing in Wind Turbines: Connecting Field Trial Results to Application Testing." The NLGI Spokesman. Mar-Apr 2019, pg. 24-33.